Fermi National Accelerator Laboratory

FERMILAB-Conf-94/430

Exergy Analysis of Tevatron Liquid Helium Satellite Refrigerator Upgrade

W. M. Soyars

Fermi National Accelerator Laboratory
P.O. Box 500, Batavia, Illinois 60510

December 1994

Presented at the *Cryogenic Engineering Conference*, June 11-14, 1991.

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EXERGY ANALYSIS OF TEVATRON LIQUID HELIUM SATELLITE REFRIGERATOR UPGRADE

W. M. Soyars

Fermi National Accelerator Laboratory*

Batavia, Illinois

ABSTRACT

Plans call for upgrading the existing Fermilab satellite refrigeration system by adding cold compressors with an accompanying phase separator return dewar. Accelerator performance will be enhanced by further lowering superconducting magnet temperatures. Two possible configurations for utilizing the stored refrigeration in the liquid dewar were studied: 1) precooling the wet expander inlet and 2) aftercooling the wet expander exhaust. A second law exergy analysis, which quantifies each source of irreversibility as a power input to the cycle, was performed to provide a comparison in operating costs between these two schemes. Aftercooling the expander exhaust results in a 1% gain in Carnot efficiency over the precooling configuration, due primarily to the more efficient use of the dewar liquid.

INTRODUCTION

The Fermilab satellite refrigeration system[1] and the Central Helium Liquefier (CHL)[2] provide cooling for approximately 1000 superconducting magnets and assorted cryogenic components that make up the Tevatron particle accelerator. The current system is capable of maintaining magnet temperatures at about 4.9K, allowing the Tevatron to operate reliably at a beam energy of 900 GeV. One aspect of Fermilab's upgrade plans involves the installation of cold compressors to further lower magnet temperatures and allow for higher beam energy[3].

The cryogenic system upgrade requires that the cold compressors, with an accompanying phase separator return dewar, be fitted into existing refrigerator building piping. The phase separator dewar protects the cold compressor from possible damage due to liquid surges and provides a source of stored refrigeration available from the liquid in the dewar. Two flow configurations were considered to utilize this stored refrigeration: 1) precooling the wet expander inlet and 2) aftercooling the wet expander exhaust. Figures 1 and 2 illustrate these configurations. Each schematic represents the layout of one of the 24 refrigerator buildings in the complete satellite refrigeration system.

To provide a thermodynamic assessment of these configurations, a second law analysis for each steady state case was performed. The second law is useful in assessing thermodynamic cycles by allowing sources of irreversibility to be quantified. An exergy analysis is a particularly useful form of second law analysis since it equates these irreversibilities to power input to the cycle, a direct measure of operating cost.

* Operated by Universities Research Association, Inc. under contract with the U.S. Department of Energy

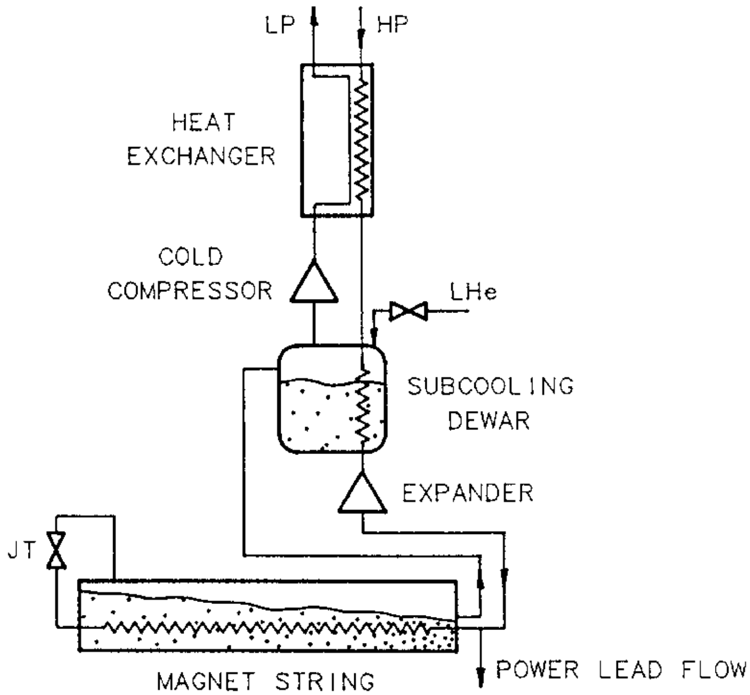

Fig. 1. Precooling Configuration

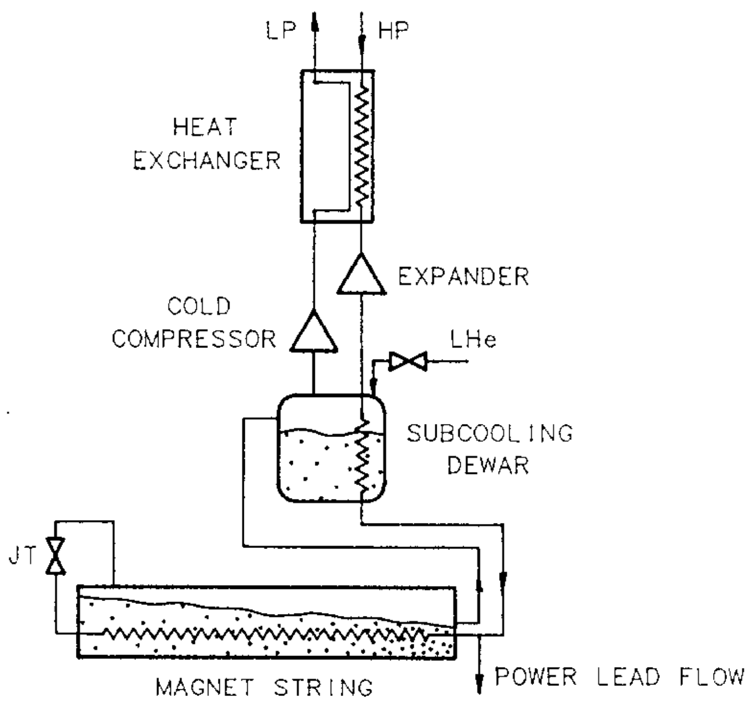

Fig. 2. Aftercooling Configuration

CYCLE CALCULATION DESCRIPTION

Satellite refrigeration performance was simulated using the assumptions and operating conditions of Table 1. Component energy balances were used to completely define state points and flow rates. The heat exchanger column was integrated into the cycle with a finite difference solution in the coldest end of the exchanger (to account for the large helium property variations in this region) and an effectiveness–NTU solution elsewhere. With all state points and flow rates known, component power requirements and heat transfer rates can be predicted.

EXERGY ANALYSIS

A popular modern method of analyzing a thermodynamic cycle is to define a quantity of "exergy" at each state point[4,5,6]. The definition of exergy uses the first and second laws simultaneously to indicate the ideal reversible work (either maximum output work available or the minimum input work required) to achieve a given state condition in a given environment:

$$W_{rev} = H_1 - H_0 - T_0(S_1 - S_0) = \text{“exergy of state 1”} = E_1 \qquad (1)$$

where 0 denotes ambient conditions of the environment, and positive work is into the system. This equation can be expressed on a rate basis as well:

$$\dot{W}_{rev} = \dot{m}[h_1 - h_0 - t_0(s_1 - s_0)] = \dot{m}e_1 \qquad (2)$$

By using exergy to express a state point in terms of work (or power) available or required, a convenient measure of value is obtained.

This concept can be extended to particular processes by considering the overall exergy change as found from "exergy balancing"[6]. By accounting for all exergy flowing into and out of some control volume, the "destroyed exergy" can be calculated. This destroyed exergy represents irreversibilities in the process which are now expressed in units of value significance, such as Watts. Then, sources of irreversibilities can be pinpointed and the cost of these irreversibilities can be easily judged. For a steady state

Table 1. Cycle Assumptions
(For typical refrigerator building in the Tevatron satellite refrigeration system)

<u>Specified operating conditions:</u>
Magnet heat load = 700W
Dewar two phase temp. = 3.5K (corresponding two phase press. = 0.46 atm)
Magnet strings return 2 phase mixture with 20% liquid to the dewar.
Precooler outlet temp. difference = 0.5K
Aftercooler outlet temp. difference = 0.1K

<u>Process assumptions:</u>

Wet expander isentropic efficiency = 75%	All single phase piping press. = 2 atm
Cold compr isentropic efficiency = 60%	All two phase piping press. = dewar press.
CHL Carnot efficiency = 17%	
Warm compr isothermal efficiency = 40%	
Total lead flow = 1 g/s	Heat exchr hi press tube inlet = 20 atm, 300K
CHL provides liquid He at 3 atm, 5.05K.	Heat exchr low press shell inlet = 1.2 atm
Single phase temp. constant through magnets.	No pressure drop across exchanger.
Magnet temp. 0.2K greater than 2 phase temp.	

process in a control volume, an exergy balance gives:

$$DE = E_{in} - E_{out} + E_W + E_Q \tag{3}$$

where DE is destroyed exergy, E_{in} is the exergy carried in by the process fluid, E_{out} is the exergy carried out be the process fluid, E_W is exergy carried in by work, and E_Q is exergy carried in by heat transfer. For an ideal process, DE equals zero; therefore, DE is always greater than or equal to zero.

Consider what is represented by E_W and E_Q and their signs. The exergy of a quantity of work is simply the work itself:

$$E_W = W \tag{4}$$

Since positive E_W is defined as exergy carried in, this term is positive when work is added. The exergy of a quantity of heat transferred is given by the Carnot work either required to produce or available from the heat transfer process:

$$E_Q = Q(T - T_0) / T \tag{5}$$

where positive Q is heat added to the control volume and T is the temperature from which Q is being transferred. When the heat transfer requires refrigeration, E_Q is negative; when the heat transfer can run a heat engine, E_Q is positive. Substituting for E_W and E_Q in (3):

$$DE = E_{in} - E_{out} + W + Q(T - T_0) / T \tag{6}$$

On a rate basis:

$$D\dot{E} = \dot{m}_{in} e_{in} - \dot{m}_{out} e_{out} + \dot{W} + \dot{Q}(T - T_0) / T \tag{7}$$

As an example, consider a case of interest in cryogenics: extracting heat of refrigeration Q from a temperature region T with $T < T_0$. Assume no work is involved. Referencing (6) shows that the E_Q term is negative and E_{in} will be greater than E_{out}. This indicates that some exergy is accounted for by the work required to operate a Carnot refrigerator extracting Q at temperature T. If the process is not doing refrigeration equal to a Carnot cycle, losses will result.

APPLYING EXERGY TO THE COLD COMPRESSOR CYCLES

Since the satellite refrigerator simulation has determined all process state points, exergy values for each point can be calculated. Then, the exergy balancing concepts can be used to compute the rate of exergy destruction in each refrigerator component. The Central Helium Liquefier is considered to be another component of the satellite refrigerator; its losses are based on the plant operating at an assumed Carnot efficiency. The rate of exergy addition to the refrigeration system is equal to the power required to run the warm and cold compressors and CHL, less the expander power produced.

Magnet refrigeration and power lead cooling results from the useful exergy remaining. For the magnet strings, this "flow" of useful exergy amounts to the Carnot power required to refrigerate the magnet heat load at the specified magnet operating temperatures. The useful exergy flow to the power leads is taken to be the exergy rate convected in by the lead flow.

Having defined for the system the total exergy input rate, the exergy destruction rate of each component, and the useful flow of exergy produced, one can generate a graphical depiction of exergy flow. When using "pie" charts, the complete pie represents the total exergy (or, equivalently, power) input to the cycle. This exergy flowing into the system is then divided into portions representing either useful exergy flows or exergy losses. The portions are identified as a percent of total power input. Note that the percentage of useful flow represents a Carnot efficiency. An ideal Carnot cycle would have all of its input exergy converted to useful exergy with no losses.

Figures 3 and 4 show the exergy flow charts for the precooling wet expander case and the aftercooling wet expander case. The aftercooling wet engine configuration is seen to be superior in efficiency. It would operate at a Carnot efficiency of about 14% while the precooler configuration would give about 13%. From a power usage standpoint, the aftercooling setup provides the same amount of cooling but requires about 8% less power. The exergy flow diagrams show that a smaller percentage of exergy is destroyed in the dewar when its used for aftercooling.

The aftercooling configuration offers the more thermodynamically efficient use of the liquid thermal buffer since it adds heat to the dewar at the lower temperature (the expander outlet temperature as opposed to the expander inlet temperature). This fact is recognizable before performing a second law analysis; however, the second law analysis allows the gains in thermodynamic efficiency to be quantified. These quantified efficiency gains from aftercooling can be more clearly judged against advantages offered by the precooling arrangement (such as slower expander speeds due to increasing inlet fluid density). Another point that the exergy flow charts reveal is that the aftercooling configuration gains some in efficiency by having a larger percentage of losses accounted for in the more efficient CHL. Finally, this analysis identifies the warm compressors as the satellite system component that generates the most inefficiency. Thus, any attempts to improve the satellite system efficiency would profit most from warm compressor efficiency improvements.

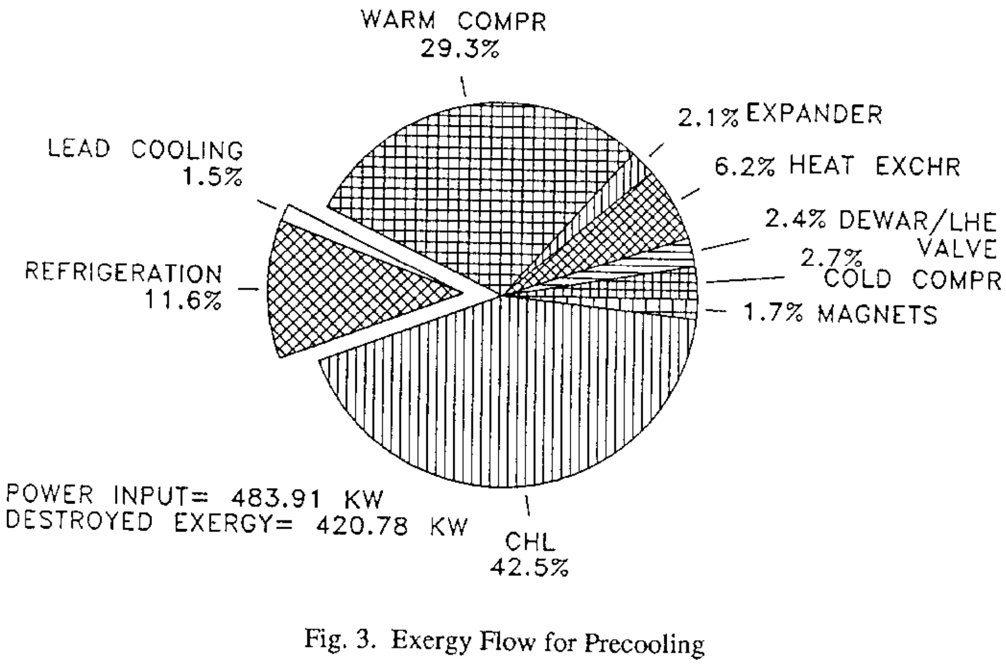

Fig. 3. Exergy Flow for Precooling

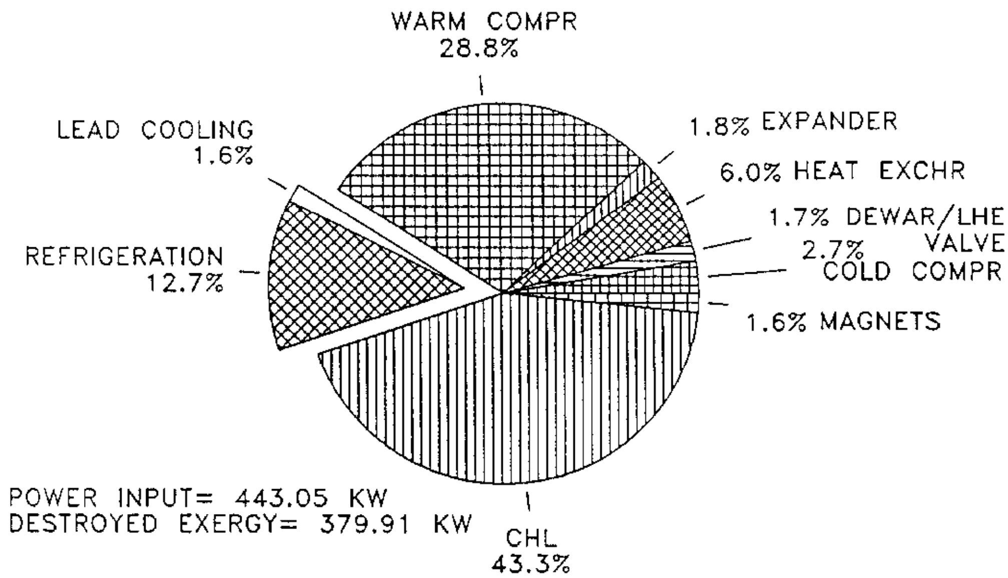

Fig. 4. Exergy Flow for Aftercooling Configuration

CONCLUSIONS

A second law—exergy analysis gives a powerful way to compare two possible refrigeration cycle configurations. Areas of irreversibilities can be identified, and a cost can be attributed to these component losses. Aftercooling the wet expander exhaust results in a 1% gain in Carnot efficiency over the precooling configuration, due primarily to the more efficient use of the dewar liquid.

ACKNOWLEDGEMENTS

The author wishes to thank Jay Theilacker for his guidance and advice in conducting this work. The help of Alex Martinez, who ran many of the computer models, was also greatly appreciated.

NOMENCLATURE

DE = destroyed exergy
E = exergy
e = exergy per unit mass
E_{in} = exergy carried into control volume
E_{out} = exergy carried out control volume
E_Q = exergy carried into control volume by heat transfer
E_W = exergy carried into control volume by work
H = enthalpy
h = enthalpy per unit mass
$\dot{m}$ = flow rate
S = entropy
s = entropy per unit mass
T = temperature
W_{rev} = ideal reversible work

Note: a dot above a quantity indicates "rate of" that quantity

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